

MODEL-BASED FUNCTIONAL PERFORMANCE TESTING OF AHU IN KISTA ENTRÉ

Pär Carling* and Per Isakson**

* *ÅF-Installation, Stockholm SWEDEN. par.carling@af.se*

** *Building Sciences KTH, Stockholm SWEDEN. per.isakson@byv.kth.se*

A seasonal functional performance test based on detailed system simulation together with intensive trending is used to commission a large AHU in the office building, Kista Entré, Sweden.

Keywords: AHU, functional performance testing, simulation, trending, commissioning.

INTRODUCTION

Kista Entré is a 50,000 m² office building in Stockholm, Sweden. It was built by Skanska Hus AB for the developer Skanska Fastigheter AB. The building was completed successively and tenants moved in during the period May 2002 to June 2003. The commissioning was not properly completed. The building owner Vasakronan AB acquired Kista Entré and took over July 1 2003. There is a two-year period covered by a guarantee, during which Vasakronan AB, ÅF-Installation AB, and Building Services Engineering at KTH perform a commissioning project based on intensive trending and performance analysis. The work reported here is part of that project.

The air-handling units (AHUs) in Kista Entré are of special design the performance of which was questioned by Vasakronan. Heat recovery from a coil in the return air to a coil in the supply air is done by a liquid-loop. Besides the two coils this loop includes heat exchangers for supply of primary heating and cooling, respectively. In the supply duct upstream the main coil there is an additional coil that recovers heat from cooling beams and thus provide free cooling (see figure 1). There are conflicts built into this system. The heat recovery by the liquid-loop is hampered by the heat recovery by the free cooling, since that increases the temperature at the air inlet of the main coil. The heat recovery by the liquid-loop is hampered also by the supply of primary heat, since it increases the temperature of the liquid-loop. The influence on the life cycle cost of these conflicts is not well understood. Last winter the heat recovery by the liquid loops of the main AHU was low due to lack of proper commissioning, an inappropriate control strategy and the design itself.

The heat recovery efficiency depends strongly on the heat capacity rate ratio between air and liquid (Incropera and DeWitt, 1996, Gudac et al. 1981, Balen et

al., 2003, Bennet et al. 1994, etc.) and fluid properties (Zeng et al., 1992). Nevertheless, the control strategy used in Kista Entré does not adapt the liquid flow rate to the airflow rates.

Holmberg (1975) has examined the optimum liquid-loop flow rate in a system with constant UA-values. He concludes that heat exchangers have maximum efficiency when the heat capacity rates are balanced ($C_{min}/C_{max}=1$). The maxima only exists for large values on the over-all number of transfer units i.e. for large coils in combination with a moderate C_{min} .

In academic research projects performance analysis based on intensive measurements and detailed simulation is established practice. However, the cost is currently high (IPMVP, 2001, www.ipmvp.org/).

Our hypothesis motivating this work is that analysis, which is supported by detailed simulation in combination with intensive trending, has a large potential in commissioning of complicated subsystem. The confrontation of the simulation model with trenddata and the subsequent calibration reveal the relevance of the model. In most cases a model that display a good enough agreement with measured data will be found. Simulation of basic cases will help the analyst to establish a deeper understanding of the performance of the subsystem. Simulation with actual trenddata as input will help the analyst decide whether the subsystem performs as it should. This might well differ from the intended behaviour. Furthermore, in the communication with other parties simulation results are helpful and complete sets of trenddata, which covers whole seasons, are much more convincing than small sets and short time series.

There are certainly economical and practical obstacles that must be overcome to realize the potential of detailed simulation and intensive trending in commissioning. We consider the approach feasible would it not require so many working hours. Thus, we anticipate there is a solution in timesaving tools and procedures. The aims of this work are:

- ✓ Quantify the cost to apply detailed simulation and intensive trending in commissioning of non-standard AHU.
- ✓ Identify measures to decrease the cost to apply detailed simulation and intensive trending.
- ✓ Quantify the value of the support we deliver to the commissioning of the AHUs in Kista Entré – both the potential and the actual support.

DESCRIPTION OF CONTROL AND TRENDING

The control of AHU maintains the set-point of the supply air temperature (T_{sd} , varies between +20 and +22°C depending on the outdoor temperature) by modulating the control valves (U_{FC} , U_C , U_{LL} and U_H). The control valves in the

liquid-loop are of on/off-type and switches operational mode, from heating to cooling, when the outdoor temperature exceeds $+17^{\circ}\text{C}$. The liquid-loop flow rate is currently controlled to be constant. The free-cooling system tries to maintain the set-point of the supply water temperature to the cooling beams. If necessary district cooling supplies additional cooling to this system. The AHU operates between 6^{00} and 21^{30} . The fan tries to maintain the set-point of the duct pressure. The air flow rates through the AHU are not balanced, more air is supplied through the supply duct than removed through the return duct. The air flow rate varies between 40 and 50 kg/s.

We sample several hundred datapoints every five (or ten) minutes via the building management system, BMS, (TAC Vista®, www2.tac.com/). All sensors are mounted and integrated in the BMS by the control contractor. The AHU, LB11, reported on here is equipped with high accuracy heat meters, which measure all flow rates and temperatures in its liquid loops. Figure 1 depicts LB11. Kista Entré comprises three buildings and there are another two AHU of the same type as LB11. The heat meters in LB11 are the only sensors that are installed because of our project. We have taken part in the practical work with the BMS and set up half of the trendlogs. We transfer data to our office over the Internet at irregular intervals. Since an appropriate application was not available the control contractor supplied us with a Java class library to communicate with their BMS over the Internet. We made a minimal application that is started interactively. We currently store data in Matlab® format (www.mathworks.com/).

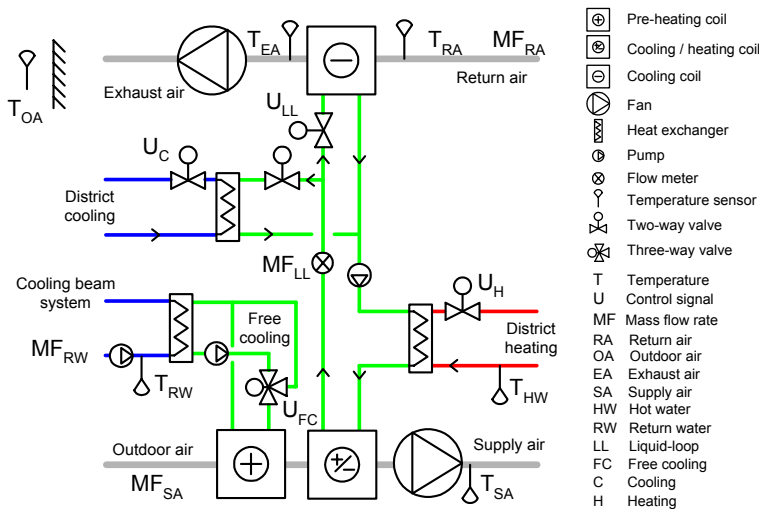


Figure 1. The main AHU LB11 in Kista Entré.

SIMULATION MODEL

We use the IDA simulation environment (Sahlin 1996, www.equa.se). IDA is a component-oriented detailed simulation programme that comprises three modules:

- ✓ A development environment. Components are described in a dedicated building description language (NMF).
- ✓ A graphical interface for compiling systems of connected components.
- ✓ An implicit differential-algebraic equation solver.

The simulation model is based on a number of assumptions:

- ✓ The heat exchange with the surrounding of the liquid-loop system and heat generation from the pump is negligible.
- ✓ No air leakage occurs within the air-handling unit.
- ✓ Heat transfer between the air streams and the connecting ducting and ambient air is negligible.
- ✓ Constant moisture content in the return and supply air respectively.
- ✓ No condensation occurs at the coil surfaces.
- ✓ Turbulent flow on both air and liquid sides of heat transfer devices.

We use the standard component library in the IDA-application Indoor climate and energy (ICE), together with two models made for this study; a heat transfer flow dependent heat exchanger and a heat transfer flow dependent coil.

The UA-value of the heat exchangers (liquid-liquid) are modelled by the equation:

$$UA = \frac{1}{k_a \cdot (m_p^{-0.8}) + k_b \cdot (m_s^{-0.8}) + k_c} \quad (1)$$

The designations are explained in the nomenclature list. The equation solver computes a UA-value for each time step depending on the flow rates. The UA-value is then used to compute an effectiveness, ε , by using the NTU-method for a counter flow heat exchanger (Incropera and DeWitt, 1996). The effectiveness is finally used to compute output temperatures.

The UA-value of the coils (liquid-air) are modelled by the equation:

$$UA = \frac{1}{k_d \cdot (m_{liq}^{-0.8}) + k_e} \quad (2)$$

The designations are explained in the nomenclature list. The equation solver computes a UA-value for each time step depending on the liquid flow rate. The UA-value is then used to compute an effectiveness, ε , by using the NTU-method

for a counter flow heat exchanger (Incropera and DeWitt, 1996). The effectiveness is finally used to compute output temperatures. It should be noted that this model do not take changes in the air flow rate into account when calculating the UA-value. In Kista Entré the airflow rate variations are moderate. No condensation is assumed which is a realistic assumption for Swedish conditions during the studied period of the year.

The control valve model is idealized; perfect linear control without hysteresis is assumed. The model contains two parameters, M_{min} and M_{max} which defines the minimum and maximum flow rates corresponding to minimum and maximum control signal, respectively.

The IDA-environment comprises a macro-function where component models may be assembled and stored as systems. In figure 2 a screen-copy of the AHU macro user interface is depicted. The upper part of figure 2 depicts the input fields for some main parameters. The lower part of figure 2 depicts the component models and their connections. It corresponds to the outline in figure 1. By double-clicking the component model icons more parameters may be specified.

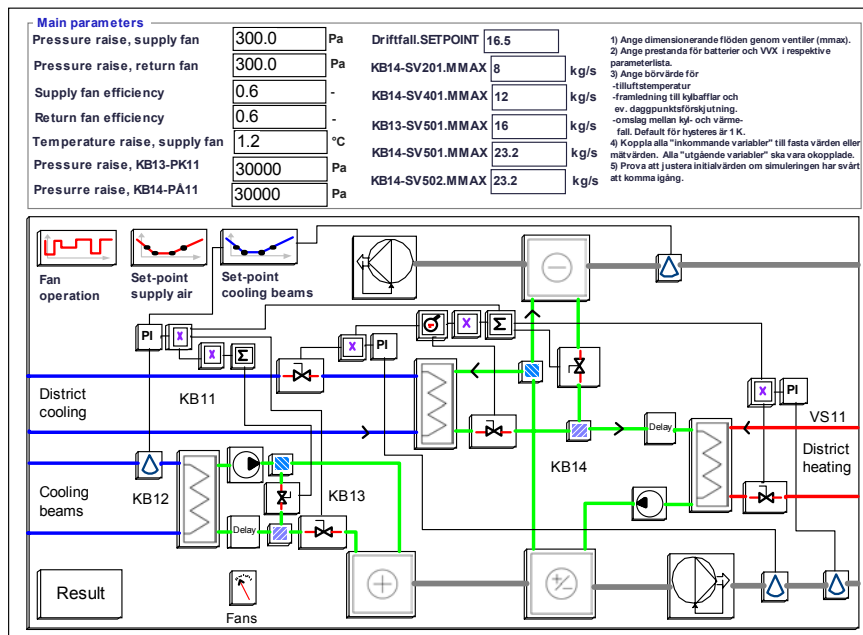


Figure 2. Screen copy of the AHU macro user interface. This picture shows the complexity of our model. Each component model is represented by a block. Many of these components are modeled with only a few lines of code.

COMPONENT MODEL CALIBRATION

In equation 1 and 2 the coefficients k_a , k_b , k_c , k_d and k_e may be calculated from catalog data or measured data. Parameter estimation techniques have been used to fit HVAC-models to data (Rabehl et. al, 1997). In the present work we tried to use data available from the design stage or from measurements in the real building. For the heat exchangers performance at one operating point was at hand. As a first approximation we thus assumed the heat resistance due to not flow-dependent conduction (including resistance due to fouling) to be very low ($k_c=0$). Since the brine solution in the loops (potassium formiat) has reasonable thermal properties, we approximated the flow-dependent resistances to be equal ($k_a=k_b$).

For the coils we used measurements to find k_d and k_e . Mean UA-values were calculated for a stable period of a few hours using the equation:

$$\overline{UA} = \frac{\overline{m_{liq}} \cdot \overline{c_{liq}} \cdot (\overline{T_{liq,1}} - \overline{T_{liq,2}})}{\overline{\Delta Tlm}} \quad (3)$$

This was done for two operating points for each coil, respectively. The liquid-loop flow rate is constant but was changed (from about 23 kg/s to 17 kg/s) at one occasion. The liquid flow rate in the free cooling system is variable.

The maximum flow rates for the different control valves (U_H and U_{FC}) were estimated from measured flow rates and either using the flow rate at fully open control valve or by extrapolating to such conditions.

The real building controller parameters are unknown. However, after simulating the system we decreased the gain parameter for all controllers from the default-value $0.3 \text{ } ^\circ\text{C}^{-1}$ to $0.1 \text{ } ^\circ\text{C}^{-1}$ in order to decrease control signal oscillations which not could be seen in real building data. It should be noted that no other adjustments of the model was done. Table 1 lists the parameters used in the component models and the source of the information.

Table1. Parameter values for components in simulation model.

Component	Parameter	Value	Source
Heat exchanger free cooling	$k_a = k_b$	7.77E-5	Catalog data and equation 1
	k_c	0	
Heat exchanger heating	$k_a = k_b$	7.54E-5	Catalog data and equation 1
	k_c	0	
Pre-heating coil	k_d	1.315E-4	Measured data and equation 2 and 3
	k_e	6.0E-6	
Heat recovery coil	k_d	3.51E-5	Measured data and equation 2 and 3
	k_e	4.9E-6	
Heating/cooling coil	k_d	3.32E-5	Measured data and equation 2 and 3
	k_e	5.0E-6	
Valve U_H	M_{max}	8.0 kg/s	Measured (extrapolated)
Valve U_{FC}	M_{max}	16.0 kg/s	Measured
Valve U_{LL}	M_{max}	23.2 kg/s	Measured

COUPLING TO MEASURED BOUNDARY CONDITIONS

The AHU-macro were coupled to measured boundary conditions; liquid-loop flow, inlet temperatures and flows. Table 2 provides information about the measured boundary conditions.

The liquid flow rate to the cooling beams was estimated from a heat balance at the heat exchanger connecting the cooling beam system and the free-cooling system. The flow rate in the free cooling system was measured and temperature sensors were available.

The airflow rate in the return and supply ducts were measured but the accuracy was poor. Instead we used heat balances at the heat recovery coil and the combined heating and cooling coil to calculate the airflow rates.

Measured data was slightly filtered to remove outliers before connecting 10-minute values to the simulation model.

The liquid-loop flow rate was averaged to 23.2 kg/s. In practice this flow is oscillating between about 21 and 25 kg/s.

Table2. Measured variables on the boundary of the AHU-macro. The variables are explained in figure 1.

System	Variable	Source	Comment
Cooling beam system	T_{RW}	Measured	
	MF_{RW}	Estimated from heat balance	No flow meter available
District heating	T_{HW}	Measured	
Liquid-loop	MF_{LL}	Measured	
Supply duct	T_{OA}	Measured	
	MF_{SA}	Estimated from heat balance	Available measurements poor
Return duct	T_{RA}	Measured	
	MF_{RA}	Estimated from heat balance	Available measurements poor

COMPARISON OF SIMULATED AND MEASURED OUTPUT

The left diagrams in figure 3 presents simulated versus measured values of the exhaust air temperature (T_{EA}), free cooling valve control signal (U_{FC}) and heating valve control signal (U_H). A valve control signal of 100% corresponds to a fully open valve. The right diagrams in figure 3 depict histogram over the difference (residual) between those simulated and measured variables. The mean value and the standard deviation are indicated in the upper right-hand corner.

Here we focus on a period during the winter 2004 which means that the cooling mode never occurs (the control valve U_C never opens). The time period is January and February 2004. The outdoor temperature (T_{OA}) is varying between -12°C and $+7^{\circ}\text{C}$. Only data for periods when plant operates is considered.

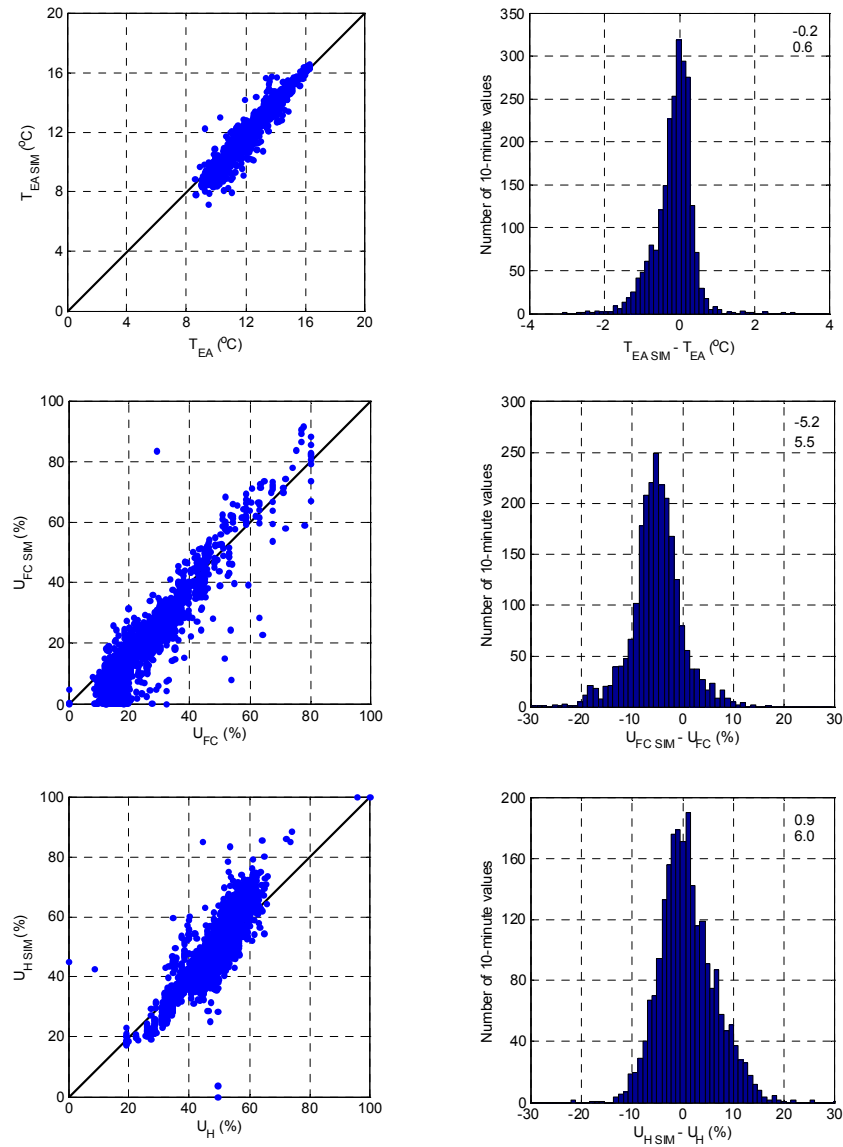


Figure 3. The left diagrams depicts simulated versus measured exhaust air temperature, free cooling and heating valve control signal. The right ones are histograms over the difference between those signals. In the upper right-hand corner the mean value and the standard deviation is indicated.

The agreement between simulated and measured output is fairly good. The liquid-loop flow rate has been oscillating which is one explanation to the scattered data. The model fails to mimic the real behavior of the free-cooling valve (U_{FC}) close to fully closed valve.

FUNCTIONAL PERFORMANCE TESTING

A performance index (PI) that is used in the contract between the contractor and the building owner in Kista Entré is a seasonal average of

$$PI = \frac{Q_{RA,LL}}{Q_{LL,SA}} \quad (4)$$

where $Q_{RA,LL}$ is the heat recovered from the return air to the liquid-loop and $Q_{LL,SA}$ is the heat transferred from the liquid-loop to the supply air. A measured average of the PI for the studied period is 0.45 and the corresponding simulated value is 0.44.

In the current system the heat capacity rates are unbalanced. Using the system model we change the liquid-loop flow rate (originally $MF_{LL}=23.2$ kg/s) so that an optimal capacity rate with respect to the return stream is obtained ($MF_{LL}=13.5$ kg/s). However optimization with respect to the supply stream generate a similar value ($MF_{LL}=13.8$ kg/s). Table 3 supplies information on measured and simulated PI's. A substantial improvement in the performance occurs when the liquid-loop flow rate is decreased so that the capacity rates are balanced.

Table3. Measured and simulated performance index defined according to equation 4. Values are averaged for operating time during the period January and February 2004.

Case	MF_{LL}	PI
Measured	23.2 kg/s	0.45
Simulated with current liquid-loop flow	23.2 kg/s	0.44
Simulated with balanced capacity rates	13.5 kg/s	0.52

It is obvious that the model may be used for further testing e.g. regarding the influence of the supply air temperature set-point on the performance index. Future studies also include using the model to explore a robust control strategy and test the model as a reference in on-going commissioning.

DISCUSSION

We are half-way through this study. The emphasis of this report is on building of the AHU simulation model and confronting it with measured data. However, we summarize the result and lessons learned so far.

Our project was out of the ordinary to the control designer and the control contractor in Kista Entré. We had problems to communicate and get acceptance for our requirements on the extra sensors. The performance of these sensors has been an on-going issue that has caused high indirect costs. Obviously, we should have paid more attention to the acquirement of the special equipment.

Setting up the trendlogs was time consuming and error prone. We had to do regular debugging. The BMS had little support for efficient management of trendlogs. We reckon that one reason for the problems is that the system is "point-oriented" and offers little overview in forms of lists or alike, e.g. it is difficult to spot an erroneous property value of a trendlog. We depend on the control manufacturers to supply better software.

Trenddata must be inspected and preprocessed before it can be used with simulation. We have done this with our own set of functions in Matlab®. These functions are developed with large sets of trenddata in mind and we think they are competitive compared to other solutions. Nevertheless, the handling of trenddata is far too costly.

The cost of making simulations largely depends on the simulation software used. There is a trade-off between modelling capability and ease of use. The simulation environment that we use (IDA) allows us to build detailed models of almost any HVAC system. We think that is essential, since modelling is most needed for systems, which are not yet predefined in simulation programs. However, we had a series of problems before the model run smoothly, which made the cost for this study high. There were problems with initial value calculation and lack of convergence, etc. We need better tools used in a more skillful manner! However, we have a reasonable experience of simulation, and thus we do not find it realistic to require higher expertise of the user. Many of our problems seems to have their origin in lack of robustness in the basic component models. A feature of IDA is that simulation models consisting of many basic component models may be encapsulated as macros, which may be saved and used in much the same way as basic component models. This macro-feature is a valuable vehicle for reuse of subsystem models.

So far we have hardly began to use our AHU-model in the commissioning of the system in Kista Entré. However, our AHU-model appears to be good enough for our purposes. After calibration it shows a fair agreement with measured data and it runs safely.

It might be premature to discuss the benefits of intensive trending and simulation in the commissioning of the air handling units in Kista Entré. Trenddata show that they do not meet the specifications regarding the heat recovery and that this has been the case throughout the heating season. Nobody

questions that. The results of our detailed simulations has raised some doubts regarding the design of the units. Furthermore, we have learned from communication with professionals, who commission similar systems, that our problems with poor heat recovery are not unique. Our contribution to the commissioning of these systems might be significant.

CONCLUSIONS

We have developed an AHU-model in the form of an IDA-macro. Using the model we were able to identify a measure that substantially increased the liquid-loop performance in the AHU we studied.

The approach generated the following conclusions:

- ✓ We have encountered more practical problems than we anticipated and because of that the cost has been unreasonable high.
- ✓ There exists problems in commissioning of this specific type of AHU that justify that we pursue our work.
- ✓ Substantial improvements of virtually all tools and procedures are needed.

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NOMENCLATURE

T	Temperature, °C.
MF	Mass flow rate, kg/s.
UA	Heat exchanger transfer coefficient, W/°C.
k_a	Flow dependent heat resistance heat exchanger, (kg/s) ^{0.8} , °C/W.
k_b	Flow dependent heat resistance heat exchanger, (kg/s) ^{0.8} , °C/W.
k_c	Heat resistance in heat exchanger material, °C/W.
k_d	Flow dependent heat resistance coil liquid side, (kg/s) ^{0.8} , °C/W.
k_e	Heat resistance on airside and coil material, °C/W.
ε	Effectiveness, —.
m	Mass flow rate, kg/s.
c	Specific heat of liquid, J/kg, °C.
M_{max}	Maximum flow rate through control valve (kg/s).
M_{min}	Minimum flow rate through control valve (kg/s).
ΔT_{lm}	Logarithmic mean temperature difference, °C.
—	Mean value.
PI	Index for the liquid loop performance.
C	Heat capacity rate (mass flow rate times specific heat), W/°C.
Q	Heat transfer rate, W.

Subscripts

OA	outdoor air
RA	return air
EA	exhaust air
SA	supply air
LL	liquid-loop
RW	return water (from cooling beam system)
CW	cold water
HW	hot water
FC	free cooling
C	cooling
H	heating
p	primary side of heat exchanger.
s	secondary side of heat exchanger.
liq	liquid side of coil.
1	supply or return liquid to coil that have the highest temperature.
2	supply or return liquid to coil that have the lowest temperature.
sim	simulated variable
min, max	heat exchanger unit with the smaller and the larger of the hot and cold fluid capacity rates, respectively.

REFERENCES

1. **Incropera F P. and DeWitt D P.** Fundamentals of heat and mass transfer. // John Wiley & Sons, Fourth Edition 1996
2. **Gudac G J., Mueller M A. and Bosch J J.** Effectiveness and pressure drop characteristics of various types of air-to-air energy recovery systems. // ASHRAE Trans., 1981 – vol.87, Part 1 - P. 199-210.
3. **Balen I., Donkerjovic P. and Galaso I.** Analysis of the coil energy recovery loop system. // International Journal of Energy Research John Wiley&Sons, 2003.- 27. - P. 363-376.
4. **Bennet I J D., Besant R W. and Schoenau G J.** A procedure for optimising coils in a run-around heat exchanger system. // ASHRAE Trans., 1994 – vol.100, Part 1 - P. 442-451.
5. **Zeng Y Y., Besant R W and Rezkallah K S.** The effect of temperature-dependent properties on the performance of run-around heat recovery systems using aqueous-glycol coupling fluids. // ASHRAE Trans., 1992 – vol.98, Part 1 - P. 551-562.
6. **Sahlin P.** Modelling and simulation methods for modular continuous systems in buildings. // Bulletin no.39. Doctoral dissertation. Department of Building Sciences. KTH Stockholm, Sweden.
7. **Rabehl R J., Beckman W A. and Mitchell J W.** Parameter estimation and the use of catalog data with TRNSYS // Building Simulation 1997.
8. **Holmberg R B.** Heat transfer in liquid-coupled indirect heat exchanger systems // Journal of heat transfer--Transactions of ASME 1975, 499-503.